Design and Realization of an Adjustable Fluid Powered Piston for an Active Air Spring

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Abstract

In this paper, we present a new compact hydraulic linear actuator. The concept is developed to change the rolling piston diameter of an active air spring during usage. By doing so, the air spring can actively apply pressure and tension forces. The actuator is designed for small movements at high forces. It is insensitive to side forces, which are introduced by the bellows rolling on the rolling piston of the air spring. A diaphragm sealing is used to minimize friction. Hence a precise adjustment of small displacements at high dynamics is possible and the system is completely leakage-free. We describe the design and development of this actuator and show first measurement results from preliminary tests to show its functionality.

KEYWORDS: active air spring, active strut, compact diaphragm hydraulic actuator, fluid powered linear actuator, side force insensitivity

1. Motivation

Within the scope of the Collaborative Research Center 805 (CRC) “Control of Uncertainty in Load-Carrying Structures in Mechanical Engineering” we develop an active air spring for employment in a vehicle. The motivation is to combine the advantages of an air spring damper, such as very good driving comfort and automatic level control, with those of an active system. Furthermore, we want to be able to handle uncertainties during operation of the vehicle. These uncertainties could vary, but common uncertainties are the vehicle load, the ride velocity, the street and the driver. In this paper, we will concentrate on the development of the active air spring itself.

The concept of the active air spring is to apply axial tension and pressure force by altering the rolling piston diameter of the air spring during usage. The axial tension and pressure forces are applied with frequencies up to 5 Hz.

2. Introduction

The aim of using active struts in general is to overcome the conflict between ride comfort and ride safety by applying forces actively. The movement of the car's body can be controlled to reduce rolling or pitching.

The only fully active suspension system available on the market is the *Magic Body Control* by *Daimler*. It is the successor of the *Active Body Control*, which was launched in 1999. A hydraulic piston in series with the steel spring applies forces with a frequency up to 5 Hz /2/. Due to the greater spring stiffness needed, the driving comfort at higher excitation frequencies is poor and the system is normally used for sports cars.

The active air spring damper combines the advantages of an air spring, such as superior driving comfort and automatic level control, with those of an active strut. The active air spring can apply axial and tension forces by altering the load-carrying area $A_{\mathrm{ax}}(t)$, which is described by

$$F_{\mathrm{ax}}(t) = A_{\mathrm{ax}}(t)(p_{\mathrm{air}} - p_{\mathrm{amb}}), \tag{1}$$

where p_{air} denotes the pressure in the air spring and p_{amb} the ambient pressure. The load-carrying area of an air spring with outside guiding of the bellows is a circle with the diameter $d_{\mathrm{ax}} = 1/2\,(d_{\mathrm{P}} + d_{\mathrm{O}})$ where d_{P} represents the piston diameter and d_{O} the outside guide diameter. The piston diameter is changed by four radially movable segments which are evenly distributed along the circumference of the piston. We use a double piston air spring as shown in **Figure 1** with a load-carrying area of the ring

$$A_{ax}(t) = A_{\mathrm{ax1}}(t) - A_{\mathrm{ax2}}(t). \tag{2}$$

The advantage is that only small changes in the diameters d_{ax1} and d_{ax2}, always in opposite directions, are needed for large relative area changes. Additionally, a coupling of the actuators of the upper and the lower pistons is possible to minimize the actuator forces needed. We have already shown the capability of such a double piston active air spring with simulations /3/. The operational capability of the active air spring with one active piston, which is shown in **Figure 2**, was already proved experimentally /1/. A mechanical transmission with camshafts, driven by a hydraulic swing motor, was used to adjust the rolling piston diameter of this first prototype.

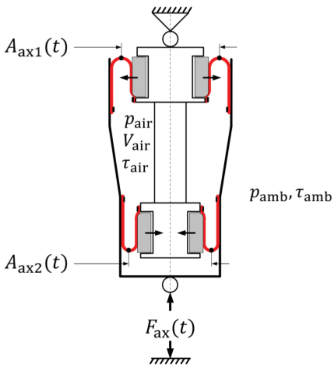

Figure 1: Schematic diagram of the double bellows active air spring

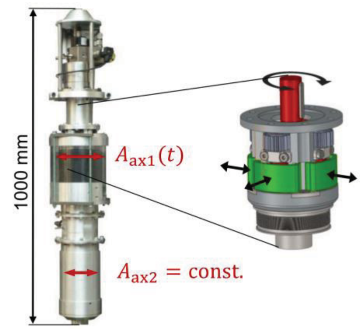

Figure 2: The first prototype of the active air spring /1/

Due to the actuator concept of the first prototype and its limitations, this active air spring can never be used in a car. The mechanical camshaft actuator and the hydraulic swing motor require almost 50% of the strut's length and are too heavy for usage in a mobile application. The actuator itself is not very efficient due to friction. An efficiency of approximately 60% was measured and mechanical wear was detected. Furthermore, no working concept for the actuator coupling could be found. Due to these limitations of the mechanical adjustment mechanism, we needed to develop a new actuator. The challenging tasks are the large actuating forces at small displacements, limited space available, the needed insensitivity to side forces and good efficiency.

3. Actuator Concepts

We applied the methods of the VDI guideline 2221 "Systematic approach to the development and design of technical systems and products" /4/ for designing the new actuator and started with the definition of the main requirements for the actuator. The required actuator displacement is $\pm 3\ \mathrm{mm}$ at an edge frequency of 5 Hz. The maximum actuator force needed is $10\ \mathrm{kN}$. A secure guiding of the segments at all times and a compensation of side forces, caused by the axial compression and extension of the air spring and the resulting rolling of the bellows on the piston (segments), which result in a maximum torque of $70\ \mathrm{Nm}$, is necessary. The available space is given by the air spring dimensions;

see **Figure 3**. Additional requirements are high efficiency and minimum wear. Furthermore, the coupling between the actuators of the upper and the lower piston needs to be easily realizable.

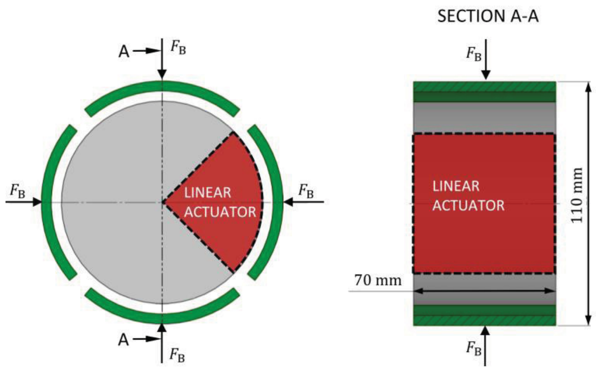

Figure 3: These sketches of the active piston show the available space for the linear actuator. The force F_B is introduced by the bellows and is dependent on the compression of the air sping

In a concept study, many actuator concepts were examined, but only two concepts seemed to be capable of fulfilling the requirements; a hydraulic actuator and a mechanical wedge gear actuator; see **Figure 4** and **Figure 5**.

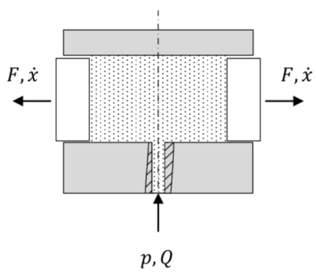

Figure 4: Schematic diagram of the hydraulic actuatur concept

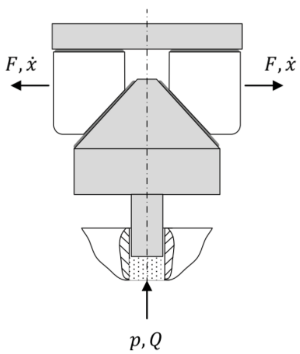

Figure 5: Schematic diagram of the mechancial actuatur concept, the wedge gear

The mechanical wedge gear actuator was already examined in preliminary experiments. The results show that the mechanical solution is not reasonable for this problem because of many problems detected during those tests, for example low efficiency and complex guiding of the segments.

4. Development of the Hydraulic Actuator

For those reasons the main focus is on the development of the hydraulic actuator. A patent search was made at the beginning of the development, but no hydraulic linear actuator was found which fulfilled our requirements. As basic design decisions we had to choose the type of hydraulic actuator and the sealing concept.

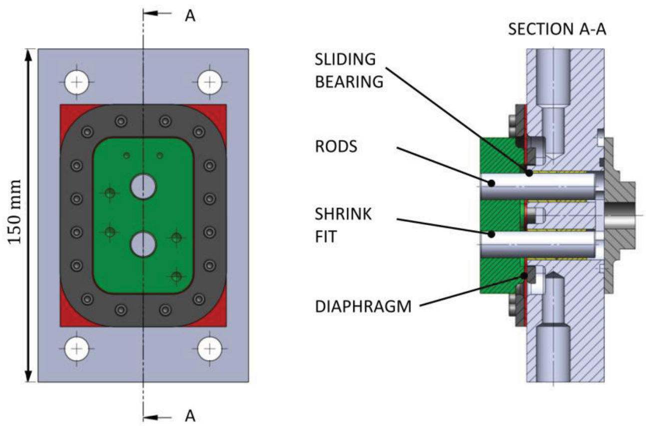

Figure 6: The hydraulic linear actuator and its components

Due to the limited space available, a double-acting linear actuator is not suitable for our problem. We use a single-acting piston, which can only apply pressure forces. The force introduced by the bellows F_{B} is used to retract the piston rod. For the sealing, three concepts are possible; a conventional piston rod seal, a diaphragm seal or a metal bellows. Two disadvantages of the piston rod seal for this concrete application are its friction due to the relative movement between the rod and the seal and its inherent leakage /5/ /6/. The friction of the seal is dependent on many operating parameters, such as relative velocity, operating temperature or operating pressure /7/. Due to an insufficient lubricating film, piston scoring can occur if piston rod seals are used for short strokes /5/. This results in a higher wear of the seal and eventually of the piston rod. For these reasons we were looking for a sealing concept where no relative movement occurred - diaphragm seals or metal bellow seals. A detailed research on metal bellow seals showed that there were no exiting solutions on the market for our application and a custom solution would be needed. On the contrary, diaphragm seals are easily available in many different versions and applications on the market. This concept has several advantages over the rod seal. It is completely leakage free and greater pressure effective areas are possible. But the main advantage is that the guiding is running completely in oil, which minimizes the friction and the wear of the rods and bearings. Due to the limited space available, two piston rods are used instead of one with a greater diameter. They are guided by linear sliding bearings; see **Figure 6**.

Based on the later actuator design for the usage in the air spring, a first prototype of the actuator is designed for preliminary tests; see **Figure 7**. The objective is to prove the actuator's functionality and to examine it experimentally. The design of the piston rods and the linear sliding bearings was optimized by applying FEM-simulations. The diaphragm was designed according to the design manual by Freudenberg /8/. It is clamped on the inside of the segment and on the outside in the main casting.

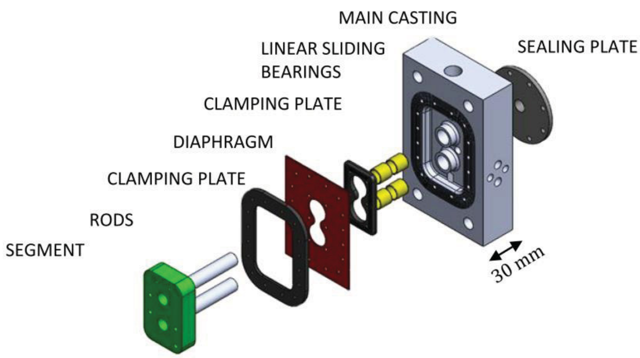

Figure 7: Design of the first hydraulic diaphragm actuator, which was examined experimentally in preliminary tests

5. Preliminary Tests of the Hydraulic Actuator

The aim of the preliminary test is on one hand to prove its functionality, e.g. insensitivity to side forces, application limits and burst pressure, and on the other hand to identify its relevant parameters such as, friction forces, displacement area and load-carrying area. The test setup is shown in **Figure 8** and the hydraulic schematic diagram in **Figure 9**. For examining the hydraulic actuator under realistic load conditions, a hydro pneumatic unit, consisting of a plunger cylinder and a gas accumulator as compliance, was used to generate a load. The static load F_0 on the actuator was changed by adapting the hydraulic pressure in the plunger cylinder. Because of the relatively small displacement volume compared with the gas volume of $0.75\ \mathrm{l}$, the load is almost constant over the entire radial travel Δr of the plunger cylinder. The proportional valve *4WRPE 6 E 32SJ–*

2X/G24K0/A1MA by *Bosch Rexroth* was used to control the displacement of the hydraulic diaphragm actuator with a simple PI-controller.

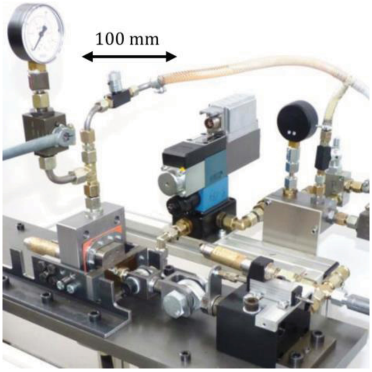

Figure 8 Test setup

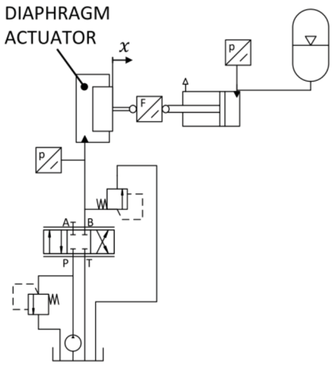

Figure 9: Hydraulic schematic diagram

For controlling the valve and for the data acquisition we used a *dSpace 1103* system. All used sensors are listed in **Table 1**. For all experiments the oil *Shell Tellus S2M 46 HLP* with a measured density $\varrho = 0.879\ kg/l$ at $40°\mathrm{C}$ is used.

Type	Name	Measuring Range
displacement	Novotechnik TS-0025	$0 \ldots 25\ \mathrm{mm}$
pressure (actuator, $p_{\mathrm{hyd}} < 30\ bar$)	Keller PA-23/8465-100	$0 \ldots 30\ \mathrm{bar\ (abs.)}$
pressure (actuator, $p_{\mathrm{hyd}} \geq 30\ bar$)	Keller PAA-33X/80794	$0 \ldots 100\ \mathrm{bar\ (abs.)}$
pressure (backpressure unit)	Keller PA-23/8465-100	$0 \ldots 100\ \mathrm{bar\ (abs.)}$
force	HBM 1C2/10kN	$0 \ldots 10\ \mathrm{kN}$

Table 1: Used sensors.

We performed various experimental measurements with different test parameters as shown in **Table 2**.

Parameter	Value
static loads (centric)	$F_0 = (2 \ldots 10)\ \mathrm{kN}$
	$T_0 = 0\ \mathrm{Nm}$
static loads (eccentric)	$F_0 = (2 \ldots 4)\ \mathrm{kN}$
	$T_0 = (33 \ldots 66)\ \mathrm{Nm}$
amplitudes in mm	$\hat{x} = (0.5 \ldots 2.5)\ \mathrm{mm}$
frequencies f in Hz	$f = (0.5 \ldots 5)\ \mathrm{Hz}$

Table 2: Test parameters.

The first functionality tests were promising. The actuator could withstand pressure of $40\ \mathrm{bar}$ without bursting. One important parameter for characterizing the actuator is the friction force F_F. It needs to be calculated from the measured quantities. One approach is to calculate it by using the equation of linear momentum for the rod of the actuator, which leads to

$$F_\mathrm{F} = (F_\mathrm{hyd} - F_\mathrm{m} - m_\mathrm{A}\,\ddot{x})/\mathrm{sgn}(\dot{x}), \tag{3}$$

F_m denotes the measured actual force, F_hyd the hydraulic force and m_A mass of the actuator. The actuator force is

$$\begin{aligned} F_\mathrm{A} &= A_\mathrm{L}\left(p_\mathrm{hyd} - p_\mathrm{amb}\right) \\ &= A_L\,\Delta p_\mathrm{hyd}, \end{aligned} \tag{4}$$

where $A_\mathrm{L} := \mathrm{d}F_\mathrm{hyd}/\mathrm{d}x$ denotes the load-carrying area of the actuator. Measurements showed that the load-carrying area is not only dependent on the displacement of the actuator as expected but also on the actuator pressure as shown in **Figure 10**. It influences the bulging of the diaphragm and therefore has a direct influence on the calculation of the friction force.

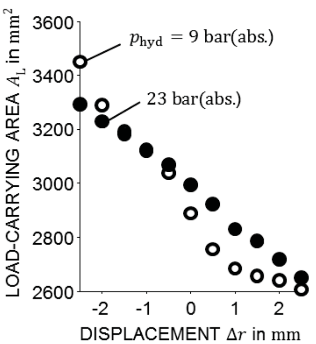

Figure 10: The measured load-carrying area A_L is dependend on the displacement of the actuator but also on the actuator pressure p_A, here $9\ \mathrm{bar}$ and $23\ \mathrm{bar}$ absolute pressure

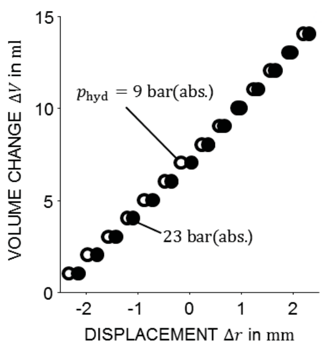

Figure 11: The change of the measured hydraulic volume ΔV over the actuator displacement, the slope is the displacement area A_D

For this reason we used another method to determine an equivalent friction force. Assuming only Coulomb friction forces and neglecting flow losses, which is suitable due to the pressure sensor, the equivalent friction force is

$$F_{\mathrm{F}} = W_D/(4\,\hat{x}), \tag{5}$$

where W_D denotes the dissipated energy per oscillation. It is the sum of the dissipated energy at the expansion and the compression of the actuator

$$W_{\mathrm{D}} = W_{\mathrm{DE}} + W_{\mathrm{DC}}. \tag{6}$$

The dissipated energy at the expansion of the rod can be calculated by subtracting the mechanical output energy from the hydraulic input energy

$$\begin{aligned} W_{\mathrm{DE}} &= (W_{\mathrm{hyd}} - W_{\mathrm{mech}})_{\mathrm{E}} \\ &= \int_0^{1/(2f)} (P_{\mathrm{hyd}} - P_{\mathrm{mech}})\,\mathrm{d}t, \end{aligned} \tag{7}$$

respectively for the compression of the rod

$$\begin{aligned} W_{\mathrm{DC}} &= \left(W_{\mathrm{mech}} - W_{\mathrm{hyd}}\right)_{\mathrm{C}} \\ &= \int_{1/(2f)}^{1/f} (P_{\mathrm{mech}} - P_{\mathrm{hyd}})\,\mathrm{d}t. \end{aligned} \tag{8}$$

The mechanical power is

$$P_{\mathrm{mech}} = F_{\mathrm{m}}\,\dot{x} \tag{9}$$

and the hydraulic actuator power

$$\begin{aligned} P_{\mathrm{hyd}} &= \Delta p_{\mathrm{hyd}}\,Q \\ &= \Delta p_{\mathrm{hyd}}\,\dot{x}\,A_{\mathrm{D},} \end{aligned} \tag{10}$$

where A_D denotes the actuator displacement area. The main advantage of this method is that the load-carrying area is not needed, rather only the displacement area $A_{\mathrm{D},}$. It is constant over the actuator travel and only slightly dependent on its pressure; see **Figure 11**. One has to mention that this is just a rough approach because the displacement area is measured in a quasi-static measurement. Nevertheless, these measurements are sufficient to show the actuator's functionality and to estimate its friction forces and efficiency. For one oscillation cycle, the efficiency is defined as

$$\begin{aligned} \eta &= \eta_{\mathrm{E}}\,\eta_C \\ &:= (W_{\mathrm{mech}}/W_{\mathrm{hyd}})_{\mathrm{E}}\,(W_{\mathrm{hyd}}/W_{\mathrm{mech}})_{\mathrm{C}}. \end{aligned} \tag{11}$$

Figure 12 shows the measured displacement, the actuator pressure and the force for one oscillation cycle. The mechanical power is calculated using equation (9) and (10).

The resulting friction force and the efficiency for the quasi-static measurement for $2\ \mathrm{mm}$ amplitude at a frequency of $0.1\ \mathrm{Hz}$ is shown in **Figure 13**. The friction force is slightly higher if the torque is additionally acting on the actuator. The efficiency plot shows an efficiency of approximately 90%.

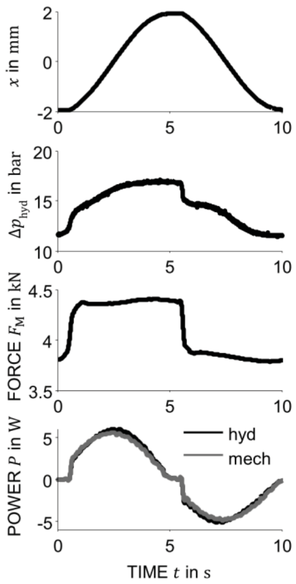

Figure 12: Quasi-static measurments at a frequency of $0.1\ \mathrm{Hz}$, an amplitude of $2\ \mathrm{mm}$ and a static load of $4\ \mathrm{kN}$

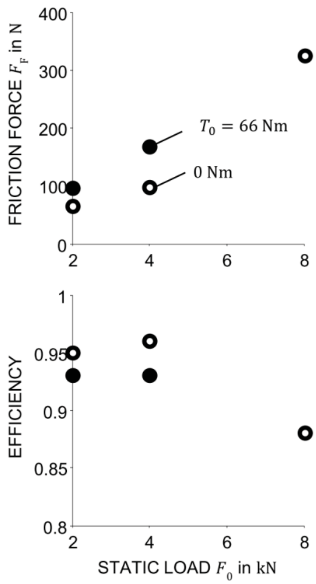

Figure 13: Friction force and efficiency for the quasi-static measurement without ($T_0 = 0\ \mathrm{Nm}$) and with side force ($T_0 = 66\ \mathrm{Nm}$)

6. Conclusion and Outlook

In this paper, we presented a new hydraulic linear actuator, which is developed to change the rolling piston diameter of an active air spring during its usage. By doing so, the air spring can apply pressure and tension forces actively. Due to the limited space available in the air spring, a compact actuator concept is needed. The main requirements are small actuator travel of approximately $3\ \mathrm{mm}$, actuator forces of $10\ \mathrm{kN}$ and insensitivity to side forces, which are introduced by the bellows rolling on the rolling piston of the air spring. We designed a compact hydraulic actuator to fulfill the requirements. A diaphragm seal is used to minimize friction. Two pistons rods, which are guided by linear sliding bearings, compensate the side forces. The guiding is running completely in oil, which minimizes the friction and the wear of the rods and bearings. The system is completely leakage-free and a precise adjustment of small displacements at high dynamics is possible.

The experimental setup to characterize the actuator will be improved in a next step to measure its parameters exactly. Furthermore, we will adapt the actuator design for utilization in the air spring. Other solutions for the sealing instead of the clamped diaphragm will be examined, such as the usage of a hose or a direct vulcanization of the diaphragm on the piston.

7. Acknowledgement

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9. Nomenclature

A_{ax}	load-carrying area of air spring	$\mathrm{mm^2}$
A_D	displacement area hydraulic actuator	$\mathrm{mm^2}$
A_L	load-carrying area hydraulic actuator	$\mathrm{mm^2}$
d_{ax}	load-carrying diameter of the air spring	mm
d_P	rolling piston diameter of air spring	mm
f	excitation frequency	Hz
F_0	static load on hydraulic actuator	N
F_{ax}	axial force of the air spring	N
F_m	measured force	N
p_{air}	air spring pressure	bar
p_{amb}	ambient pressure	bar
p_{hyd}	hydraulic actuator pressure	bar
P_{hyd}	hydraulic power	W
P_{mech}	mechanical power	W
Q	volume flow	$\mathrm{m^3/s}$
x	actuator displacement	mm
T_0	static torque acting on the hydraulic actuator	Nm
τ_{air}	air temperature in air spring	K
V_{air}	volume of air spring	l
W_D	dissipated energy	J